CHAPTER 7

Longitudinal Strength

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CHAPTER 7

Longitudinal Strength

Introduction

1 Following discussion in previous chapters of prediction of loads, of material selection and of initial estimation of structural parameters, it is now necessary to synthesise the structure to match the earlier estimates and to analyze its strength in resisting the applied loads. To begin, the primary wave bending moments will be considered as these dominate longitudinal scantlings, and it is the purpose of this chapter to discuss means of designing the structure to provide the necessary longitudinal strength. The design methods covered assume that all the structure is effective in resisting wave loading; there will, however, in any complete structure be areas, as defined briefly in Chapter 2, which cannot take the full load because they are by-passed by the load transmission paths, for example close to large holes or near the ends of short decks. Methods for estimating the efficiency of these areas of structure in taking load are given in Chapter 12.

2 The process to be followed for design of primary structure is straightforward although somewhat time consuming as inevitably it will involve frequent iteration. Because the methods are not always simple to describe in words, they will be illustrated through the chapter by a worked example, and the same example will also be used in later chapters when discussing design of more detailed aspects. Any design process inevitably involves the pursuit of many red herrings up blind alleys! It would be intolerable to the reader if all these excursions were to be included, as well as making the chapter unnecessarily lengthy, but during such pursuits new and better starting points for the next stage frequently become apparent. Consequently it is not feasible to present the design example in a form whereby the reader can recalculate all the steps himself, as there are a number of apparent non-sequiturs, or points at which data may seem to be missing. In any case, the purpose of the example is not to give the reader an excuse to check the author's arithmetic, but to demonstrate all the *essential* steps in the design process, and by means of the summary at the end of the chapter to ensure that nothing is omitted in a real design. It should also be emphasised that the design example is not taken to a final conclusion because in the later stages the analysis becomes repetitive as the structure is refined. Furthermore, because for any particular design there is likely to be a requirement for optimisation of the structure (as discussed in Chapter 11) to minimise cost or weight, or to maximise reliability, the latter parts of the design process could follow one of a number of different paths, inclusion of which would confuse the current presentation. Nevertheless, a viable solution to the design requirement is presented.

Design Example (1)

Introduction and Initial Estimates

3 It will be assumed that a procedure similar to that presented in Chapter 6 has been followed and the designer has derived a simplified structural section as shown in *Figure 6.3* with the overall dimensions being as follows:

Length (WL)	120 m
Beam (WL)	15 m
Draught	4 m
Depth	10 m (no effective superstructure)
Submerged Area	50 m^2
Flare	5°
Design Bending Moment	400 MN.m (sagging) 320 MN.m (hogging)
Design Shear Force	20 MN

The section is taken to be one of a number of critical sections along the length of the hull and has principal dimensions as illustrated in Figure 7.1. For realism, a second deck will be assumed at a height of 7.5 m above base, and both 1 and 2 decks will be penetrated by a hole symmetrical about the centreline of half width 1.5 m.

4 Omission of an effective superstructure is deliberate. If the superstructure is effective, the *longitudinal* parts of it are designed in exactly the same way as for decks and longitudinal bulkheads, and so to include it here would be unnecessarily repetitive. The design of a deckhouse which is *not* considered effective, but which still has a significant load carrying efficiency, is a different matter and will be covered in Chapter 8.

5 The structure will be built of steel to NES791 Part 2 which, in the thicknesses likely to be used, has a yield strength (σ_y) of 280 MPa (see *Table 5.1*). To estimate the fraction of yield stress that should be used during this early stage of design, consideration should be given to acceptable values of the ratios b/t and a/k where a and b are the stiffener spacings respectively transversely and longitudinally, t is the plate thickness and k is the radius of gyration of a longitudinal stiffener and width b of plating. Smith (1983) recommends that plate slenderness ratio ß ($=(b/t)\sqrt{(\sigma_y/E)}$) should ideally be less than 2.0 and never more than 2.5, and that column slenderness ratio λ ($= (a/\pi k)\sqrt{(\sigma_y/E)}$) should normally be less than 0.45 and never more than 0.55. However, for the value of σ_y quoted above these recommendations give values of $b/t<68$ and $a/k<47$, suggesting that, for example, for 8 mm plate b should not exceed 544 mm. This is considered to be too restrictive at this stage of the design as it would lead to heavy plate if wider stiffener spacing was desired, and so a range of b/t between 60 and 90, and of a/k between 40 and 70 is recommended. Any design limitations that may result from a choice of too high a slenderness ratio will become apparent in the detail analysis later, and can be corrected then if necessary. For this range of b/t, therefore, and from *Table 6.2*, it follows that the deck stress should not exceed 0.57 σ_y (160 MPa) and the keel stress not exceed 0.43 σ_y (120 MPa) so then the required section moduli (Z), using

$Z = M/\sigma$, will be as given in *Table 7.1*. The tensile values are inserted for completeness but are of little significance because a far higher fraction of yield may be accepted when buckling cannot occur.

TABLE 7.1
Required Section Moduli (m^3)

	Hogging	*Sagging*
Deck	+2.00	-2.50
Keel	-2.67	+3.33

6 If it is assumed as a first approximation that the neutral axis is at mid depth then the minimum acceptable second moment of area (I) is half hull depth multiplied by the highest compressive value of Z from *Table 7.1*, which gives 13.4 m^4. There are a number of ways of proceeding from this point, for example a rectangular box section may be assumed, but the simplest means of obtaining a first shot at a mean thickness is to assume that all the effective material is concentrated in the deck and bottom. Hence, to provide the required section modulus, an equivalent thickness of steel of 18 mm would be needed. It must be emphasised that this is only a guide to a first estimate of the material necessary, but it is clearly a help to have a starting point. Noting that the sides are also contributing, a first attempt at the complete section might be as shown in *Table 7.2*; the letters refer to the points marked on *Figure 7.1*. The thicknesses are derived from assuming the 18 mm required from the above estimate are now continued around the sides in reasonable proportions (from experience), with those parts subjected to lateral pressure and furthest from the neutral axis being the thickest.

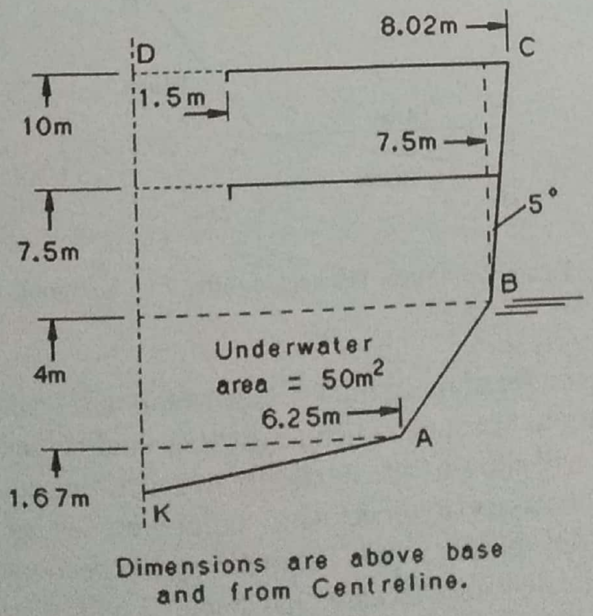

Figure 7.1 Dimensions of Example Section

TABLE 7.2
First Attempt at Mean Plate Thickness

Item	*Thickness*
CE	15 mm
BC	13 mm
AB	14 mm
KA	16 mm

7 The moduli derived from these figures are 2.52 m^3 for the deck and 1.97 m^3 for the keel; clearly, while the deck is just sufficient, the keel is not. However, there is now a new starting point and after various adjustments the structure in *Figure 7.2* was assembled, taking proper account of the internal deck and of the holes in the decks. The moduli are now 2.82 m^3 for the deck and 2.97 m^3 for the keel which, with reference to the requirements of *Table 7.1* for *compression*, is about 10% more than enough. It should be noted also that it has been necessary to insert a double bottom to provide a reasonable balance in the material thicknesses (that is to avoid a very thick outer bottom) and to maintain a neutral axis near half depth. The inclusion of the double bottom will, of course, have implications well beyond those of structure, for example arrangement and volumes of tanks, and support to machinery. Such considerations are outside the scope of this book but would in a real design have to be reviewed in depth and the best compromise solution reached.

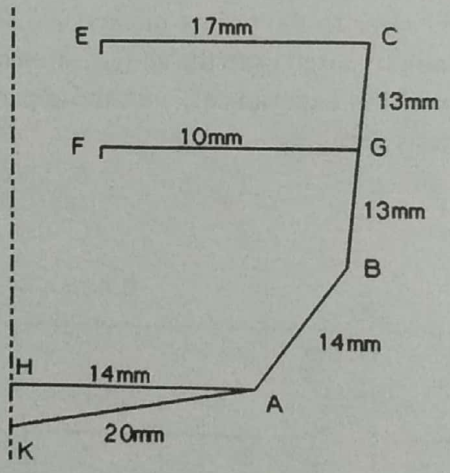

Figure 7.2 Mean Thicknesses after Two Iterations

8 Having put together a set of mean thicknesses which will lead to acceptable compressive stresses it is now necessary to fit actual structure to these thicknesses; the small margin of 10% referred to in paragraph 7 will allow a little flexibility in generating scantlings. In considering the structure required it is necessary to understand failure modes as well as likely structural imperfections. Up to now only elastic simple beam theory has been used, but the actual failure of the structure under the ultimate loads discussed in Chapter 3 will be well beyond the elastic limit and, for conventional orthogonally stiffened plated structure in compression, will be of a complex elasto-plastic form.

Compressive Failure

9 Failure modes have been discussed briefly in Chapter 2, but in relation to the collapse of plated grillages a more detailed explanation is presented by Faulkner in Chapter 21 of Harvey Evans (1975). For a perfect structure with material following an ideal elastic-plastic stress/strain curve there are three successive modes of collapse to consider, namely buckling of plating between stiffeners, buckling of longitudinal stiffeners and plating between transverse stiffeners, and overall buckling of the grillage between stiff supports (usually, for a deck, bulkheads and the ship's sides). The first of these modes will not constitute ultimate failure in a well designed structure as, even when the plate has buckled, there will still be a region of plate along the edge which is carrying load and assisting the longitudinal stiffener to resist buckling. For a method of design to resist this mode of failure, should it be of interest, see Soares (1988) which includes an allowance for initial imperfections. When the longitudinal stiffeners buckle there will be no further *local* reserve of strength and that particular panel will collapse, shedding load onto adjacent panels. If the adjacent structure is not too close to collapse and can absorb the extra load, then the whole structure may still have some reserve of strength, but usually it will be inefficient to design elements of the structure with large differences in load bearing capacity, so at this stage of design it is common to assume that ultimate collapse occurs when any one grillage collapses. This is supported in practice by the fact that the critical areas for collapse will be the upper deck and the bottom which, with fairly uniform loading and scantlings, will give little overall reserve of strength after stiffener failure. Estimation of the load for complete hull failure will be discussed later (paragraphs 67 et seq). Note, however, that in stiffened *fibre reinforced plastic* structures there are other failure modes which can interact with each other; these special failure modes are discussed by Smith and Dow (1987).

10 There is an additional mode of failure known as 'stiffener tripping' which involves sideways collapse of the stiffener rotating about its toe with associated lateral bending of the web. This mode, which can be caused by both in-plane and lateral load, is generally guarded against by ensuring that the proportions of the stiffeners are adequate, as indicated later (paragraph 19).

11 A real structure will, however, be neither perfectly flat nor will it have a zero stress state to start with. It will be initially distorted due to construction processes such as welding while at the same time there will be residual stresses present due to cold forming and temperature changes during welding. Furthermore, if the builder tries to straighten the structure to reduce initial distortions then additional residual stresses will be caused, and there is no practical way of avoiding the problem, although it can be minimised by very careful assembly sequences. The structural designer therefore has to take account of these imperfections in estimating the collapse load of the as-built structure which will be significantly less than the ideal.

12 This reduced mean stress at failure is mainly associated with Von Karman (1924) (there are earlier references - see Faulkner (1975) for an historical survey) who presents a formulation for the 'effective width' (b_e) of plate alongside each stiffener which can reach yield stress at failure while the remainder of the plate is effectively unloaded. As indicated in *Figure 7.3*, the average stress (σ_{pa}) in the plate at failure is equal to $\sigma_y b_e/b$, and much of the later work revolves

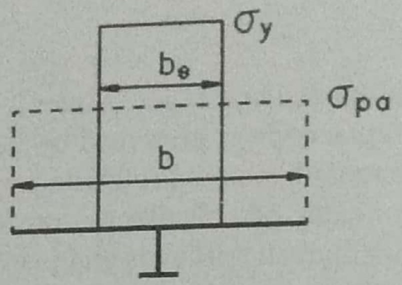

Figure 7.3 Effective Width of Plate

round methods of estimating the value of b_e at different levels of loading up to collapse. Von Karman proposed

(7.1) $$b_e/t = 1.9\sqrt{(E/\sigma y)}$$

with the yield stress being replaced by the actual edge stress for values below collapse. This formulation was in reasonable agreement with contemporary experimental evidence but was based mainly on slender steel and aluminium aircraft structures and does not necessarily apply to heavier marine structures with lower values of b/t. More recently Faulkner (1975) undertook a broad survey of available tests and proposed

(7.2) $$b_e/b = 2/\beta - 1/\beta^2$$

where

(7.3) $$\beta = b/t\sqrt{(\sigma_y/E)}$$

This formula is also valid for a state before collapse if σ_y is replaced by the plate edge stress.

13 A study of the literature will reveal a number of other proposals but that which is most valuable is due to the work of Smith et al (1988) which is based on numerical analysis of a variety of plate configurations using a non-linear finite element method and validating the results against much existing and some new experimental work. The results are presented in the form of 'load shortening curves', a typical set of which is shown in *Figure 7.4* but those which should be used in design, taken from the reference, are included for convenience as data sheets (*Figures 18.3-18.5*) in Chapter 18. These load shortening curves, which are in the form of normalised stress/strain curves over the whole elastic-plastic tension to compression failure range, can be used to obtain the value of b_e using the expression

(7.4) $$b_e/b = E_s/E = \sigma/E\varepsilon$$

where E_s is the secant modulus (the ratio of stress to strain at any point along a non-linear stress/strain curve) and σ and ε are the stress and equivalent strain taken at the appropriate point along the curve. The curves are presented for different values of β for small, average and severe initial imperfections, based on the definitions of Somerville et al (1977) for warships and of Antoniou

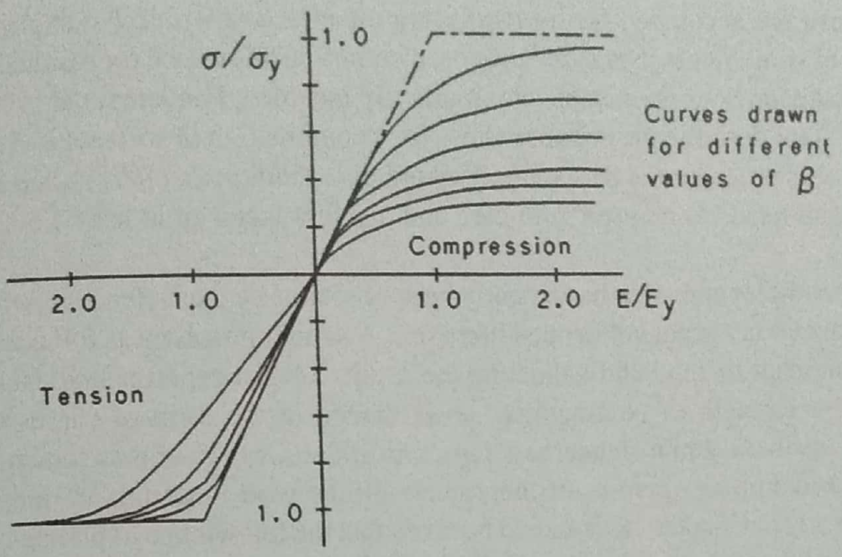

Figure 7.4 Typical Load Shortening Curves

(1980) and Antoniou et al (1984) for merchant vessels. Unless there is evidence to suggest otherwise it is recommended that the *average* value curves are used in *design*. It should be noted that the curves given in Chapter 18 are for values of a/b 1; for square plates and plates loaded in the transverse direction, as well as for biaxially loaded plates, the necessary design data can be obtained from Smith et al (1988).

14 Once a knowledge of the effective width of the plate is available it is possible to estimate the collapse stress of longitudinal stiffeners using column buckling theory. Traditionally a designer would have used the Euler column formula up to yield, conservatively assuming pinned ends. However, this takes no account of residual stresses or initial lack of straightness and so a large safety factor would have been necessary. Faulkner et al (1973) present a review of various formulations for estimating the elasto-plastic buckling strength of beam-columns. They conclude that the so-called Johnson parabola, based on column slenderness ratio λ, is adequate for ship type structures with the limit of proportionality of the steel as-built being about half the yield stress, where

(7.5) $$\lambda = (a/\pi k)\sqrt{(\sigma_y/E)}$$

That is

(7.6a) $$\sigma_c/\sigma_y = 1 - \lambda^2/4 \quad \text{for } \lambda < \sqrt{2}$$

and

(7.6b) $$\sigma_c/\sigma_y = 1/\lambda^2 \quad \text{for } \lambda = \sqrt{2}$$

where σ_c is the column collapse stress. Note, however, that the radius of gyration k in equation 7.5 is related to the *effective* width of plate for the area, and a 'reduced' effective width for the inertia.

Residual stresses are accounted for by modifying the effective width. No explicit allowance is made for lack of straightness, but as the proposed empirical allowance for residual stress is based on experimental data, both phenomena are implicitly included. Faulkner et al (1973) propose an iterative method for deriving the collapse stress for a combination of stiffener and effective width of plating, but this procedure is less soundly based than Faulkner's (1975) plate effective width formula (7.2) and needs to be used with care and a safety factor of at least 1.5.

15 A much simpler approach has recently been proposed by the Defence Research Agency at Dunfermline, but which is not yet in open literature. A similar process was followed as for plates, using a finite element method and validating the results against experimental data. it was found that the collapse strength of columns can be presented in the form of curves of normalised collapse stress against column slenderness ratio for different values of ß as shown in *Figure 7.5*. Once again, the definitive versions of these curves to be used in design are presented as data sheets (18.9-18.11) in Chapter 18. It should be noted that the full width b of plating is to be used in the calculation of ß and λ, and that the weakening effects of imperfections and residual stresses are allowed for in the curves which again are given for small, average and large imperfections. Unless there are good reasons to the contrary, average imperfections should again be used for design. For convenience, as will be seen later, the data on these sheets are also presented in an inverted form giving curves of σ_a/σ_y against ß for different values of λ (sheets 18.12-18.14).

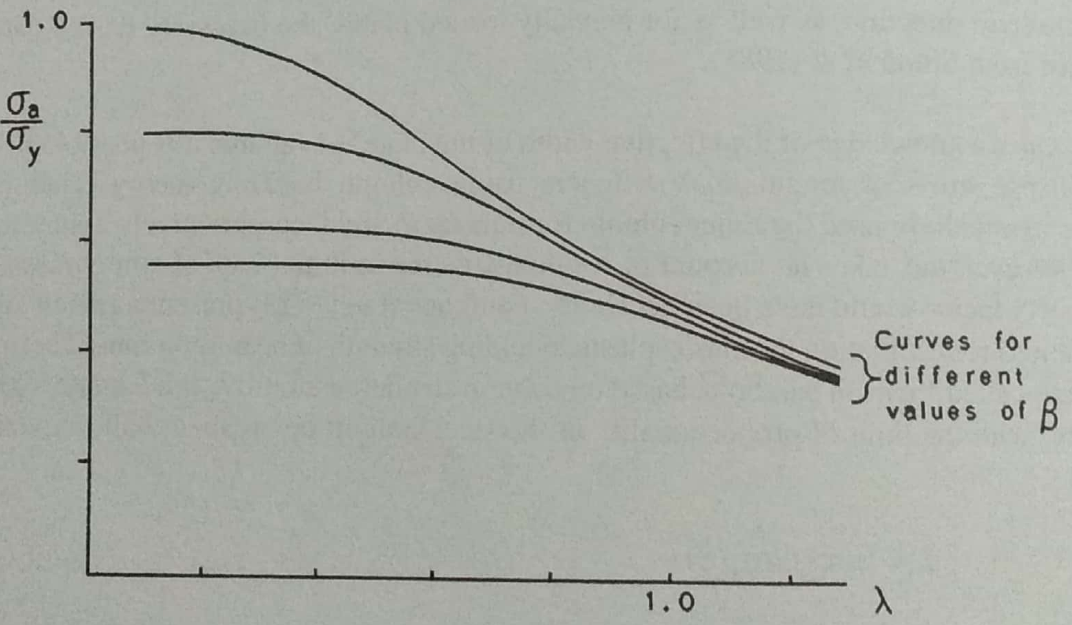

Figure 7.5 Typical Column Collapse Curves

16 Using these curves there is clearly no longer a need to iterate to obtain the collapse strength for any particular plate/stiffener combinations; it is simply a matter of calculating the nominal values of λ and ß using the full plate width b and, for the particular value of λ, interpolating between the curves for the correct value of ß and reading off σ_a/σ_y. σ_a is then the average stress across the plate and stiffener combination at collapse and equates to σ_{pa} in the Von Karman/Faulkner approach. If the critical stress at the stiffeners (σ_c) is required, it can be obtained from equation 7.7, having first found the effective width of plate (b_e) from the plate load-shortening curve (data sheets 18.3-18.8).

(7.7) $$\sigma_c = \sigma_a (A_s + bt)/(A_s + b_e t)$$

where A_s is the nominal cross section area of the stiffener.

17 Referring to the modes of failure listed in paragraph 9 is remains to check that the overall buckling strength of the grillage is acceptable and that stiffener tripping will not occur. The first is not a straightforward process, especially if proper allowance is made for imperfections, and a full non-linear finite element analysis should be undertaken. However, if the plate and column buckling calculations have been adequately carried out, and the transverse stiffeners properly sized to withstand lateral loads (see paragraph 23) then it is very unlikely that overall grillage buckling will be a critical problem. It will be sufficient to use an approximate method to check the strength, and a suitable one from the same stable as those presented above for plate and columns is given by Clarke (1979). As this again is not in the open literature, the design curves and an explanation of their use are given in Chapter 18 (data sheets 18.5-18.29). This method, incidentally, also allows for consideration of lateral loads in conjunction with in-plane loads.

18 Stiffener tripping, or torsional buckling, is a difficult phenomenon to predict accurately as it depends strongly on, amongst other things, the rotational restraint at the toe of the web and the lateral bending stiffness of the web. However, a theory derived from energy considerations is presented by Bleich (1952) covering elastic buckling due to both in-plane and lateral loads, and this is developed by Faulkner et al (1973) for the **in-plane load case** making some allowance for rotational restraint at the toe. It can be shown that the critical mode for practical Tee section stiffened structures is usually a single half wave of deflection, and an estimate of the critical elastic buckling stress (σ_{tr}) can therefore be derived as follows:

(7.9) $$\sigma_{tr} = (\sigma_s A_s + \sigma_p bt)/(A_s + bt)$$

with

(7.7a) $$\sigma_s = GJ/I_o + \pi^2 E\,(I_z d^2 + \Gamma)/I_o a^2$$

and

(7.8b) $$\sigma_p = \frac{\pi^2 E t^2}{12\,(1 - \nu^2)\,b^2}\left(\frac{a}{b} + \frac{b}{a}\right)^2$$

where σ_s and σ_p are terms allowing separately for stiffener and plate buckling effects,

and $E =$ Young's modulus
$G =$ Shear modulus
$\nu =$ Poisson's ratio
$A_f =$ Area of flange
$A_w =$ Area of web

A_s = Total stiffener area = $A_f + A_w$
I_z = Second moment about axis through web
I_o = Polar second moment about toe
J = Torsion constant = $(t_w{}^3 d_w + t_f{}^3 d_f)/3$ for a Tee section
Γ = Warping constant = $A_f{}^3/144 + A_w{}^3/36$ for a Tee section
(see also Murray and Attard (1987))
d = Distance from toe to shear centre (height of section for Tee)
a = Stiffener length
b = Stiffener spacing
t = Plating thickness.

From checks against what little experimental evidence is available it is recommended that σ_{tr} should not be less than $1.3\sigma_c$ (that is the stiffener *critical* stress, not the *average* stress). For other types of stiffener, particularly flat bar and bulb flats (OBPs), buckling may not be in a single half wave, as discussed by Smith and Kirkwood (1983), and it may be desirable to carry out a more detailed analysis if the nominal value of σ_{tr} is less than, say, three times σ_c. Methods of analysis, especially appropriate for flat bar stiffeners, are presented by Caridis and Frieze (1988 and 1989) and by Bhat and Wierzbicki (1989).

19 In evaluating equation 7.8 it should be noted that all the terms are positive and so for realistic scantlings it will usually only be necessary to calculate σ_p and the first term of σ_s to ensure that the tripping stress is adequately high. For large fabricated stiffeners it will usually be sufficient for good resistance to tripping to ensure that

$$d_f/t_f \leqslant \sqrt{(E/\sigma_y)} \tag{7.9a}$$

$$d_w/t_w \leqslant 1.5\sqrt{(E/\sigma_y)} \tag{7.9b}$$

and

$$a/d_f \leqslant \sqrt{(E/\sigma_y)} \tag{7.9c}$$

where d_f = Flange height t_f = Flange thickness
d_w = Web breadth t_w = Web thickness

Design Against Lateral Load

20 Although the subject of this chapter is longitudinal strength it is not possible in all cases to separate the longitudinal and transverse effects and so it is necessary to spend a little time in considering the effects of lateral loads which, in this context, will always be manifested as quasi-static pressures.

21 The effect of pressure on a plated grillage can be divided into the same three areas as for in-plane load, that is, on the plate between stiffeners, on a stiffener and associated plating and on the complete grillage. It is not usually necessary, however, to be concerned with the plating on its own because even if yielding occurs the resulting out-of-plane deflections will bring membrane forces into play and the reserve of strength is very large. No instabilities are involved except in the case where there is an initial lack of flatness towards the applied pressure; the effect of the pressure may then be to 'snap' the panel through which can cause an irritating noise in service and may lead to local fatigue cracking. However, it is unusual for such panels to continue snapping through for a long time as they will quickly take up a stable deflection in the opposite sense; nevertheless it is desirable to straighten markedly distorted panels during hull construction; tolerances will normally be provided in the ship specification.

22 It is not strictly possible to analyse an isolated length of stiffener from a grillage as the adjacent stiffeners will affect the boundary constraints, in particular in bending and torsion. However, as the stiffeners in one direction are usually significantly stiffer in a typical ship grillage than those in the orthogonal direction, it is acceptable when undertaking the initial sizing of the members to treat the stiffer ones first over the full span between supporting structure, and then to size the less stiff members over a span between the stiffer members. This method is not particularly accurate, however, and more details on analysis of grillages under lateral pressure are given in Chapter 10.

23 In carrying out the initial sizing, a stiffer member is then treated as a simple beam ignoring the existence of the less stiff members. Clamped ends may be assumed if the pressure is applied over a large area simultaneously so that each beam length is deflected in the same direction, for example on underwater structure, but pinned ends should be assumed if the pressure is transient and may only be applied to one length at a time so that the adjacent length may deflect towards the pressure, for example on deck structure. The plating will not deflect evenly with the stiffener but will be affected by shear lag (see Chapter 2) and the effective breadth to be taken with the stiffener should be whichever is the least of a/2 or b/2 (depending on direction) and 40t. At the initial stage the margin against yield using the loads from Chapters 3 and 4 should be not less than 1.2 but may be lower if the load is rare and well defined, for example, a test pressure. When sizing the less stiff beams, the same procedure may be used with the effective breadth acting with the stiffener again taken as the less of a/2 or b/2 and 40t but, conservatively, it is always safer to use pinned boundaries at the synthesis stage whatever the actual boundary conditions. Because of the additional distortion caused to the less stiff members by the deflection of the stiffer ones, a higher margin against yield is desirable, say about 1.5.

24 Bending is not the only condition to be considered although it will usually be the one which drives the design. It is also necessary to take account of shear loads, which may be estimated from the total load on the area in question spread over the webs of the stiffeners that are resisting the load. Using the crudest estimate of shear stress (load/web area) it is sufficient at this stage to ensure that the applied stress is less than half the shear yield stress (which itself is commonly taken as half the tensile yield stress if better information is lacking). In sizing the stiffeners, however, remember that the smaller stiffeners must pass through the larger ones involving a cut-out in the web, and sufficient web material must remain to provide adequate shear strength, as well as sufficient space between the flanges for welding.

25 The final mode of failure which is rare but which still must be checked is that of stiffener tripping. The following formula for the critical tripping stress (σ_{tc}) is derived from that presented by Bleich (1952) but modified for typical ship arrangements under **lateral load**:

$$\sigma_{tc} = \frac{GJ}{I_o} + E\,(I_z d^2 + \Gamma)\,\frac{\pi^2 n^2}{I_o a^2} + \frac{Ea^2}{I_o n^2 \pi^2 \,(3b/t^3 + 3.64 d_w/t_w^3)} \tag{7.10}$$

where G = Shear modulus
E = Young's modulus
J = Torsion constant = $(t_w^3 d_w + t_f^3 d_f)/3$ for a Tee section
I_o = Polar second moment of area about toe (excluding plate)
I_z = Second moment of area about axis through web (excluding plate)
Γ = Warping constant = $A_f^3/144 + A_w^3/36$ for a Tee section (see also Murray and Attard (1987))
A_f = Flange area
A_w = Web area
d = Distance from toe to shear centre = d_w for a Tee section
d_w = Web height
t_w = Web thickness
d_f = Flange width
t_f = Flange thickness
a = Stiffener length
t = Plate thickness
b = Stiffener spacing.

When the flange is in tension it is sufficient to take n = 1 but with the flange in compression the critical tripping stress (σ_{tc}) has to be minimised with n. The applied stress is that in the stiffener flange derived from simple beam theory as discussed in paragraph 23. As equation 7.10 only takes account of elastic instability and is derived from energy considerations it does not give a particularly reliable estimate and a margin of a factor of at least 4 between the applied and critical stress is recommended in primary structure, although in less important structure the factor may be reduced perhaps to 3.

26 In using equation 7.10 it should be noted that all the terms are positive and so for practical section profiles it will often be possible to establish the safety of the scantlings from the first term alone, only minimising the other two terms if there is still some doubt. To avoid repetition it should also be noted that the first two terms (with n = 1) are the same as those in equation 7.8a.

Shear Strength

27 In addition to the in-plane and lateral loads, shear loads need also to be considered in the design of hull sections. Shear stresses due to the wave induced shear forces will be a maximum in vertical or near vertical structure, which is in the main the ship's sides, and so before proceeding to demonstrate the design, therefore, it is necessary to discuss the estimation of the shear strength of such structure.

28 The initial effect of shear on a plated grillage is to cause shear buckling of the plate between stiffeners, which may happen at very low stresses if there is significant out-of-flatness of the plate. However, it is of little importance as diagonal tension fields immediately form to take the load, and providing the stiffeners are adequate there is no risk. Having said that, if the shear load is cyclic as it usually will be on a ship, the tension field will snap from one diagonal to the other, which can be noisy and may also precipitate fatigue failure, so it is still best avoided! The critical elastic shear buckling stress (τ_c) for a perfectly flat panel of plate is given by the expression

$$\tau_c = \chi E\,(t/b)^2 \qquad (7.11)$$

where χ is derived for different aspect ratios and boundary conditions from data sheet 18.37 which is itself taken from ESDU data sheet 71005 (1976). τ_c is then reduced by a plasticity factor (η_c) which is taken from data sheet 18.38 (from the same reference) and described in paragraph 30 of Chapter 18.

29 For longitudinal strength purposes it will usually be sufficient to ensure that the value of τ_c so derived is greater than the applied design shear stress (τ_d) from the beam formula

$$t_d = FA\bar{z}/It \qquad (7.12)$$

where F = The applied shear force at the section
$A\bar{z}$ = The moment of area about the neutral axis of the material between the outer fibre and point at which the shear stress is being calculated
I = The second moment of the whole section about the neutral axis
t = Plate thickness.

30 Shear buckling of long panels stiffened in the short direction only, for example vertical strips of side plating between frames, constitutes the next level of complexity. The addition of stiffeners increases the stress at which the panel will buckle but the information available, from aeronautical sources, can only cover equally spaced stiffeners where the web of the stiffener is at least as thick as the plate. Thus standard long stalk Tees are strictly outside the scope of the method and may fail by local buckling of the web before the combined failure mode is reached. However, for lack of better information the following method will have to be used but it is recommended that the critical buckling stress (τ_{sc}) is at least twice the design shear stress from equation 7.12 at the stiffener nearest to the neutral axis.

31 τ_{sc} is found as follows from data sheets 18.39 for clamped edges and 18.40 for pinned edges; it should be noted that the data sheets only cover the cases for open section stiffeners with negligible torsional stiffness and of the same Young's modulus as the plate. For other cases it will be necessary to use the ESDU sheet 02.03.02 (1983) from which the data sheets in this publication are derived. Curves are given for the value of K in the equation

$$\tau_{sc} = KE(t/b)^2 \qquad (7.13)$$

for different values of the aspect ratio a/b and a stiffness parameter μ given by

(7.13a) $\mu = \sqrt{(I_{st}b/a^2t^3)}$

where I_{st} is the second moment of area of the stiffener alone about its toe and the other variables have the same meanings as before. It is assumed that Poisson's ratio is 0.3.

32 It is not easy to make recommendations on whether to assume pinned or clamped edge conditions. Considerations of symmetry suggest the use of clamped edges as the stiffeners will try to take up the same deflected form in each adjacent panel, but in any doubtful case it is safer to assume pinned edges.

33 There is no available theory for overall shear buckling of an orthogonally stiffened grillage but neither is there any evidence of overall shear failure ever having occurred in a ship. It would seem, therefore, that provided the unidirectionally stiffened panel criterion above is met, and that the frames are sized adequately to withstand lateral pressure loads and are significantly stiffer than the longitudinals, then overall buckling need not be considered.

Design Example (2)

Upper Deck

34 The point has now been reached when the design example can be continued using the methods described in the foregoing paragraphs to develop the scantlings of the decks, bottom and sides. In *Figure 7.2* initial estimates of mean thickness were given based on simple beam theory to provide about 10% more section modulus than was actually needed. These thicknesses dictate the weight of the effective longitudinal structure so there is little scope for weight reduction except within the 10% margin. The aim of the development of scantlings should be, in the first instance, to provide a *safe* structure which at later stages can be refined to minimise cost, weight or some other parameter. At the same time the different elements of the structure should be consistent in terms of stiffener spacing while avoiding large steps in stiffness between adjacent elements (which are undesirable both for avoidance of fatigue failure and resistance to explosive loading - see Chapters 13 and 15).

35 It will be assumed that the structure of the example is to be stiffened with 'Admiralty Long Stalk Tee Bars'. These sections were developed in the 1950s to provide a well balanced symmetrical section with good bending stiffness and good resistance to tripping out of plane. Properties of these sections are given in data sheet 18.1; it will be seen that the range is limited and it will frequently be difficult to match the actual strength required, with the consequent need for overmatching and some unnecessary weight. The alternative, used extensively in main structure of merchant ships, is 'bulb flats' (offset bulb plates or OBPs), details of which are given in data sheet 18.2 taken from BS4848 Part 5. The range is very much greater and it is much easier to obtain a good match to the required strength. However, the sections are asymmetrical and will trip more readily under both end and lateral load than will symmetrical sections. Smith et al (1984) have shown that to achieve comparable collapse strength the weight of OBPs may be as much as

8% greater than the equivalent Tee bar and so the advantage of the greater range is not so clear cut. It is suggested that OBPs are not considered as replacements for Tee bars to save weight unless the weight of Tee necessary is more than 5% above the ideal. OBPs are, however, cheaper than Tee bars weight for weight due to the larger quantities rolled and the less complex rolls required. The balance between them is not an easy one to strike, but mixing the two types should be avoided, as connecting bulk sections to Tee sections is frequently difficult and leads to a structure in which alignment and good detail are more critical than is desirable in practical ship construction.

36 Smith et al (1984) also suggest that flat bar stiffeners be considered. These have the advantage of extreme cheapness, being cut from plate or strip, while still being symmetrical. To avoid premature tripping a height/thickness ratio of less than ten is recommended, and there will clearly be a weight penalty when compared with Tee sections. However, the simplicity of flat bars should lead to considerable saving in construction labour cost as well as some material cost saving. Perhaps the best compromise structure is to use Tee sections where weight is critical, but a combination of large Tees with intercostal flat bars in less weight sensitive areas.

37 Returning to the upper deck of the design example the aim will be to achieve a simple structure which should be cheap to produce without being overly heavy. This implies fairly widely spaced stiffeners with the majority of the weight (75-85%) in the plate. As a further guide to the starting point (but *not* a critical requirement) the web thickness of the stiffener should be approximately half the plate thickness. Given then from *Figure 7.2* a mean thickness of 17 mm, try type 3 Tees with a cross sectional area of 1613 mm^2 and a web thickness of 6.4 mm. The implication is plate 13 mm thick (76% of 17 mm as well as twice web thickness) but to achieve the required total area 16 stiffeners are needed with a spacing (rounded) of 400 mm. This is rather close and will lead to more than necessary welding with associated distortion and difficulties in maintaining form. As a second attempt try type 5 Tees with an area of 2606 mm^2 and web thickness of 7.4 mm; this with 14 mm of plate will require 800 mm stiffener spacing and, with 15 mm plate, 1.3 m spacing. However, plate only exists in round numbers of millimetre thicknesses, and it is necessary to deduct rolling and corrosion tolerances from the nominal thickness which will be for this thickness and position in the ship 1.3 mm and zero respectively (see Chapter 5). Taking then type 5 Tees at 1.0 m spacing with a plate thickness of 13.7 mm (15 mm nominal) gives a mean thickness of 16.3 mm which is less than that required but within the 10% margin (paragraph 34), while at the same time meeting the requirements for a simple robust structure.

38 From the modulus in paragraph 7 and the design sagging bending moment of 400 MN.m, a compressive stress of 142 MPa will be applied but, as the thickness is slightly less than taken at that stage, an applied stress of 150 MPa will be assumed ($\sigma_a/\sigma_y = 0.54$). Reference to data sheet 18.10 for $\beta = 2.68$ and $\sigma_a/\sigma_y = 0.54$ shows that λ should not be more than about 0.8. The radius of gyration k for the proposed combination of stiffener and plate is 53.6 mm and so, in round figures, the maximum acceptable span is 3.5 m. It will be noted, however, from data sheet 18.10 that the stress curves are very flat for this combination of λ and β and so such a high frame spacing may be risky, especially when considering the effects of lateral pressure later. The frame spacing will arbitrarily therefore be limited to 2.5 m for the present, giving $\lambda = 0.546$ and $\sigma_a/\sigma_y = 0.575$, that is σ_a is still only 161 MPa, which confirms the flatness of the stress curve and does not constitute a large margin at this stage.

39 The value of a/k is 47 and so both it and b/t are within the acceptable range for conventional structure (paragraph 5).

40 Before proceeding with other elements of the structure it is necessary to check the deck panel against overall buckling and to do this the size of the transverse deck beams is needed, which will be dictated by the requirements of lateral pressure. The upper deck design pressure in the absence of a deckhouse (the worst condition) is given in Chapter 3 as 25 kPa and suitable margins taking account of the likelihood of this pressure being exceeded and the crudity of the calculation will be 1.2 against yield in bending and 2.0 against shear yield.

41 For a span of 6.52 m (that is, from the deck edge to the edge of the hole, assuming adequate longitudinal support at that point) and beam spacing of 2.5 m the load/unit length will be 62.5 kN/m and for pin-jointed ends (appropriate for the upper deck as the pressure may be localised and adjacent panels may deflect in the opposite sense) the maximum bending moment at mid span is 332 kN.m, and the required section modulus for stiffener plus effective breadth of plate is 1423 cm^3. The effective thickness of the plate is 13.7 mm and the equivalent breadth of plate is 40t (<a/2) or 550 mm. The largest rolled Tee section from data sheet 18.1 will be insufficient and so it will be necessary to fabricate a larger one or to reduce the beam spacing. The former solution is more desirable as the design is being aimed at a minimum of stiffeners and the stiffener crossings, and it can be shown that a section with a web of 350 × 10 mm and a flange of 150 × 22 mm will have a section modulus, including the effective breadth of plate, of 1437 cm^3 which is just more than sufficient. At the same time, if *only the webs of the beams* are considered, the applied shear stress is 58.2 MPa which is well below the half shear yield criterion.

42 Returning to overall buckling, a full analysis would require a non-linear finite element method, but for an accuracy sufficient at this stage of design the method due to Clarke (1979) can be used, as discussed in paragraph 16. For the dimensions now arrived at the non-dimensional line load N_y^* is equal to -0.637 and the flexibility ratio μ is 105.1 (N_y^* and μ are defined in paragraph 14 of Chapter 18) so referring to data sheet 18.17 it is clear that the deflection due to the in-plane load is negligible and that the grillage is still stable. The deflection is, in fact, so small that it will not be necessary to use the later data sheets to calculate the extra secondary bending stresses as described in Chapter 18.

43 A check on tripping under in-plane loads as described in paragraph 18 shows that $\sigma_{tr} = 371$ MPa, which is well above the recommended minimum. Tripping can also occur under lateral load as described in paragraph 25 below, but in this case the nominal critical stress with the flange in tension is about 2600 MPa!

Bottom Structure

44 The bottom structure in *Figure 7.2* is highly idealised and before further design work can be undertaken some additional detailing is necessary. A practical double bottom will have both a keel and some vertical floors and these have been added, as shown in *Figure 7.6*; the vertical floors are positioned at 1 m and 3 m from the centre line and would, in practice, be arranged to fit machinery and tankage requirements; the keel is made 1 m high and 0.5 m wide with the flange a nominal 20 mm thick, the web 15 mm and the flat keel 20 mm. For the purposes of this exercise

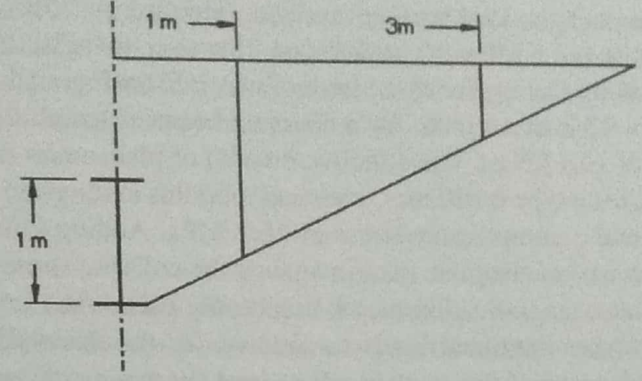

Figure 7.6 More Detailed Bottom Structure

the dimensions are typical and would be refined at a later stage with particular reference to docking loads, as specified in Chapter 4.

45 There are now three elements of the bottom longitudinal structure to be designed; the outer bottom, the vertical floors and the tank top. Work at Cardiff University reported by Evans et al (1984), and with further theory presented by Vilnay (1982 and 1983) and by Narayanan (1983), provides a method for *analysing* a double bottom structure as a single entity, including cut-outs in the webs discussed by Evans and Shanmugan (1979), by idealising the structure as a deep grillage. A discussion of this method is, however, presented in Chapter 12 as it cannot be used directly as part of *design* as required here. For design it is reasonable, and indeed necessary, to treat the elements of the structure separately and the process will be demonstrated in the following paragraphs. Should the vessel being designed have only a single bottom, then the procedure to be followed is virtually identical to that described for the outer bottom alone.

Outer Bottom

46 From the dimensions in *Figure 7.2* it can be deduced that there is a total of 216,900 mm² in the half bottom structure and deducting the material in the keel (paragraph 44), allowing the recommended rolling and corrosion margins, leaves 201,575 mm² for the inner and outer bottom and the floors. A fair distribution of steel between these elements would be mean thickness of 12 mm for the inner bottom, 10 mm for the floors and 17 mm for the outer bottom, the logic for the distribution being that 10 mm is probably about the minimum that can be accepted for structure so far from the neutral axis, and 12 mm the same but with the additional requirement to withstand some lateral pressure. The 17 mm remaining is not unreasonable for an outer bottom which has to withstand both in-plane load and lateral pressure simultaneously.

47 The floors are 2 m apart so if a single longitudinal stiffener is fitted to the bottom between them a type 6 Tee would lead to plate greater than 13.8 mm thick, allowing for the 10% margin as discussed in paragraph 34, that is 16 mm nominal or 14.2 mm after deduction for rolling and corrosion margins, which here are 1.3 mm and 0.5 mm respectively as proposed in Chapter 15. Using the same process as described for the upper deck structure this combination of stiffener and plate would then have an average collapse stress of 166.9 MPa. Taking the hogging bending

moment of 320 MN.m and the keel section modulus (reduced by 10%) from paragraph 7, an applied *primary* stress of 118.5 MPa can be deduced. However, there is, in addition, a *secondary* pressure load on the bottom plating due to the head of water. From Figure 3.10 the design pressure head will be 7.29 m or 73.3 kPa which, for a clamped beam of length 2.5 m, gives a bending moment at midlength of 19.1 kN.m. The effective breadth of plate under lateral load will be 500 mm (b/2) which gives, for a type 6 stiffener, a section modulus at the plate surface (compression surface) of 1178 cm^3 and a compressive stress of 16.2 MPa. Adding this to the primary stress gives 134.7 MPa which has an adequate margin against the collapse stress above, without being too inefficient. Note that a magnification factor, commonly used when investigating the *elastic* buckling of columns under combined loads to account for the destabilising effect of lateral pressure, is *not* required at part of this method where load shortening curves are used to estimate collapse stress. The critical stress derived from the column load shortening is directly applicable whatever combination of loads is actually causing the stress at the compression face, provided that the in-plane load predominates. The failure prediction becomes more conservative as the proportion of stress due to lateral pressure increases.

Inner Bottom

48 The inner bottom or tank top is closer to the neutral axis and the applied primary compressive stress is only 77.6 MPa (including the 10% margin). However, it will be assumed that the double bottom space is indeed a tank and will be subject to a test pressure giving a design water head of 10 m (100.6 kPa). (Note that, in the actual design, the design head will need careful consideration to allow for accidental overpressurisation or for fatigue effects.) This pressure is likely to dominate the design and initial scantlings will need to be sized accordingly. As a first attempt, assuming a 1 m longitudinal stiffener spacing and a 2.5 m span, the maximum bending moment for a clamped beam is 52.4 kN.m. It is clearly not desirable for the structure to yield under the test pressure but, on the other hand, the pressure is only applied occasionally and independently of other loads so a small margin, say 10%, against yield to allow for accidents or uncertainties is reasonable. This then would require a section modulus for stiffener and associated plating of 206 cm^3, and reference to data sheets 18.30-18.36 will show that for the assumed stiffener spacing and plate thickness no standard Tee less than type 5 is enough. To provide a mean thickness of 12 mm (as described in paragraph 46) a nominal 11 mm plate would be required giving a b/t of 109 which is too high for efficient design and for the initial assumptions (paragraph 5).

49 The simplest solution, without increasing weight, is to halve the stiffener spacing to 500 mm (note that it is desirable to keep to a fraction, or indeed a multiple, of the standard spacing that has developed so as to simplify integration with bulkhead stiffeners in due course). Halving the spacing halves the required section modulus to 103 cm^3 and this can be achieved using a type 3 Tee on 10 mm nominal thickness plating ($Z = 124$ cm^3 from data sheet 18.32) while at the same time providing the required equivalent thickness. An in-plane buckling check shows that the average buckling strength of the combination is 184.2 MPa which should be adequate, but a check on combined in-plane and pressure loading would be necessary when the pressure load is finally defined.

50 A check against stiffener tripping under later load using equation 7.10 shows a critical tripping stress of 3880 MPa which is well above four times the applied bending stress, so this failure mode is very unlikely; similarly for in-plane load the critical tripping stress from equation 7.8 is 1150 MPa which is adequately higher than the stiffener in-plane critical stress of 198.0 MPa.

Vertical Floors

51 The height of the inboard floor is about 1.5 m and so 12 mm nominal thickness plate on its own has far too high a b/t value; however, the equivalent thickness can be met by a single type 2 Tee at half height and 10 mm nominal thickness plate. The average buckling stress of this combination is 104 MPa (b/t is still quite high so it is not particularly efficient, but for these elements of the structure the aim is simply to provide the required cross sectional area while avoiding premature buckling). The applied stress at the height of the stiffener is 95.9 MPa and so in the absence of any pressure load the structure is adequate (clearly pressure load would need to be considered if the floor was part of a tank boundary). The critical tripping stress under in-plane load is 335 MPa which compared with a value of σ_c of 131 MPa is more than sufficient.

52 The distance between the inner and outer bottom at the outboard floor will be less, at about 1.0 m, but there is little to be gained from creating a separate design and so a similar structure to the inboard floor will be assumed, with the stiffener at half height.

Internal Deck

53 This deck is close to the neutral axis and so the applied stress will be low; from the equivalent thicknesses shown in Figure 7.2 it will be about 72.9 MPa, so it may be assumed that the scantlings will be dominated by the lateral pressure. From Chapter 4 the deck will be taken to have a requirement to support a high density compartment, and so the pressure load will be 15 kPa.

54 For a transverse beam spacing of 2.5 m and a longitudinal girder spacing of 1.0 m the maximum bending moment on each girder, assuming built-in ends, is 7.8 kN.m, and designing up to yield stress (as the loading is conservative) means that a section modulus at the flange of 27.9 cm^3 will be required. Reference to data sheets 18.31 and 18.30 shows that a Type 2 Tee will be more than adequate with any thickness of plate, while a type 1 is always too small. There is, therefore, a case for considering flat bar girders as suggested in paragraph 32. Further, as this deck is light in any case there is little point in precisely matching the mean thickness of 10 mm from *Figure 7.2* and the design may be aimed at minimum acceptable strength.

55 While keeping to a girder spacing of 1 m a short numerical study shows that 8 mm plate with flat bars of 90 mm height and 12 mm nominal thickness (10.7 mm after deduction of tolerances) give a section modulus at the outer fibre of the bar of 29.2 cm^3 which is sufficient for the pressure load. However, b/t is 149 and a/k is 134, both of which are well outside the preferred range and a rather inefficient structure should result. Although well off the end of the data sheet curves, the average collapse stress should be about 70 MPa, which is a little below the applied stress. However, the inclusion of minor structure means that it is unlikely there will be a completely unsupported span of 2.5 m, and in any case this deck makes only a small contribution to overall longitudinal strength, so it will be best to leave the arrangement for now and return to it when a more detailed layout and better information on supported weights are available.

Side Structure

56 The side structure can be divided into three areas as indicated in *Figure 7.2*, that is the sheer strake (CG), the mid side strake (GB) and the lower side strake (BA). The first of these has shear combined with primary bending stresses and the last has shear combined with some primary bending and sea pressure, so it is simpler to start with the mid side strake.

Mid Side Strake

57 Considering first the spacing of longitudinals, the overall girth from point A to point G is just over 7 m and it will be desirable to have a heavy stiffener (say a type 7 Tee) at about the waterline, that is about 3.5 m below G, to withstand berthing loads - the actual size of this stiffener is a minor detail which can be covered later when considering berthing loads. If 1 m spacing is used then this stiffener will be 0.5 m above or below the nominal waterline, but as the waterline is likely to rise as the ship grows in weight a little above is preferable so 1 m spacing is not unreasonable. The mean thickness of the plating from *Figure 7.2* is 13 mm which, allowing for the 10% margin, suggests type 4 Tee stiffeners on 12 mm nominal thickness plate.

58 For this arrangement it can be shown, using equation 7.11, that $\tau_c = 118$ MPa, while from equation 7.12 at the worst height in the plate $\tau_d = 100$ MPa so the plate shear strength is satisfactory. However, from equation 7.13 the panel buckling strength is only 126 MPa which is well below the required margin of 2.0 on τ_d. After some trial and error, and noting that the value of τ_d increases as the neutral axis is approached, it can be shown that it is necessary to reduce the spacing to 700 mm to achieve an acceptable structure. A study of data sheet 18.39 reveals that for this configuration only b and t have any appreciable effect on shear buckling strength, and reducing b is preferable to going to larger and perhaps unrealistic values of thickness. However, in this case it is possible to retain adequate strength with type 3 Tee stiffeners and with 700 mm spacing the deep waterline stringer can be placed on the current waterline or one space above, whichever is preferred.

59 A check on lateral sea pressure using the value of 50 kPa proposed in Chapter 3 leads to a bending stress of 154 MPa which has a margin of more than 1.2 below yield and is, therefore, acceptable, and the nominal shear stress in the stiffener webs is only 28.6 MPa.

Lower Side Strake

60 This strake will actually have significant curvature, although it is assumed flat in the idealised section shape. The assumption of flatness is conservative for in-plane buckling, and the effect of curvature may be taken as a bonus at this stage. The in-plane load on the lower side strake is similar to that on the strake above and so as a first approximation the same scantlings may be chosen. The difference between the lower side strake and the one above is that the former is subjected to rather higher lateral pressure, being below the nominal waterline. However, the top of the strake has very similar shear loading to the strake above and it is unlikely that lighter scantlings will be acceptable, and this is borne out by calculation. Checking against a lateral pressure at the bottom of the strake of 56.3 kPa (Chapter 3) the tensile bending stress is 186 MPa, which is acceptable, but now the compressive bending stress in the plate must be considered alongside the direct stress due to hull bending.

61 The hull bending compressive stress on the lowest stiffener is 52.6 MPa and the secondary compressive stress due to hydrostatic pressure is 76.3 MPa giving a total of 129 MPa. As shear loads are also present it is recommended that the factor between the applied compressive stress (σ_{dc}) and the average direct collapse stress of the plate/stiffener combination (derived from an unpublished MOD source) is given by:

$$\sigma_a/\sigma_{dc} \geqslant \tau_{sc}^2/(\tau_{sc}^2 - \tau_d^2) \tag{7.14}$$

where τ_{sc} is defined by equation 7.13 and τ_d is the applied shear stress at the stiffener in question. In this case $\sigma_a = 152.4$ MPa, $\tau_{sc} = 233$ MPa and $\tau_d = 90$ MPa so the condition is just met.

Sheer Strake

62 The height of the panels is 2.5 m and *Figure 7.2* requires an effective 13 mm thickness (less 10%) which would lead to either a single type 4 Tee at half height on 12 mm plate or two type 3 Tees at 833 mm spacing. The first has a b/t which is far too high so the second is the best start. Investigating plate shear, the applied shear stress in the lower plate section is 119 MPa while $\tau_c =$ 139 MPa so b/t could be slightly higher. A check on panel buckling gives a value of $\tau_{sc} = 172$ MPa while τ_d at the lower stiffener is 83 MPa, so again the factor of 2 is achieved.

63 The applied direct stress due to hull bending is 132 MPa and, unfortunately, this cannot be met with a type 3 stiffener; in fact the stiffener has to be increased to type 5 to give an adequate value of σ_a of 154.0 MPa. Checks on other factors such as combined local bending, shear and direct stress show this combination to be just acceptable although it is a little heavier than that originally intended.

Final Structure

64 The structure resulting from the foregoing work is illustrated in *Figure 7.7* with the shape faired while still having the same underwater properties, as specified in paragraph 3; although for the purposes of this example it has been termed 'final structure', it is in reality only the end of the first iteration around the section and more work would be necessary to refine it to the actual design requirement. The properties of the section are given in *Table 7.3* with the original properties being aimed at from *Figure 7.2* in brackets. As a guide to the success of the synthesis process, *Table 7.4* compares the applied average stresses on the different parts of the structure with the average collapse stresses.

65 The difficulty in achieving full bending efficiency will be noted from *Table 7.3* where the weight has increased above the initial estimate by 4% while the bending strength has reduced by 6%, although these are both well within the original 10% margins that were allowed. The next

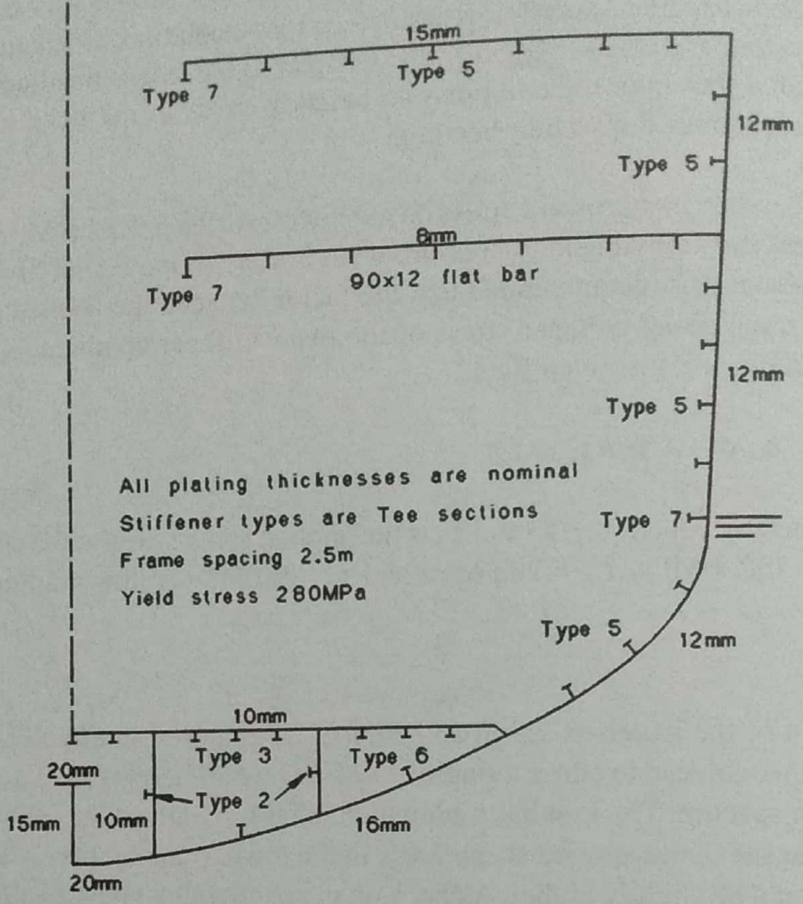

Figure 7.7 First Full Definition of Final Structure

TABLE 7.3
Properties of Section in Figure 7.7

Weight/unit length (T/m)	8.41	(8.09)
Neutral axis (NA) above base (m)	4.92	(4.86)
Second moment of area about NA (m^4)	13.65	(14.47)
Upper deck modulus (m^3)	2.69	(2.82)
Keel modulus (m^3)	2.77	(2.97)

(Original properties from *Figure* 7.2 in brackets)

stage could be to try to reduce weight, but it must be noted that from the initial assumptions in *Figure 7.2* that that can only be done by reducing the equivalent thicknesses and *increasing* the applied stress. While there is some freedom to design grillage structures to withstand higher applied stresses than those considered above it will be increasingly difficult to do so with a *reduced* equivalent thickness, and the effect will be to reduce plate thickness while increasing the

TABLE 7.4
Margins Against Initial Estimates of Compressive Collapse for the Structure in Figure 7.7

	Applied compressive stress (MPa)	Collapse stress (MPa)	Margin
Upper deck	149.1	161.0	1.08
Shear strake	125.7	154.0	1.23
Internal deck	75.8	70.0	0.92
Ship's side above WL	52.2	160.7	3.08
Ship's side below WL	128.7 #2	160.7	1.25
Inner bottom #1	76.0	184.2	2.42
Outer bottom	147.5 #2	166.9	1.13
Vertical floors	93.5	104.0	1.11

#1 Margin will allow for tank working pressure (undefined as yet).
#2 Includes secondary compressive bending stress.

number of stiffeners and considerably increasing the cost and complexity of the structure. To put some limit on the exercise it must be noted that from data sheet 18.10 extremely low values of λ and ß are required to raise the stiffener collapse stress above 80% of yield and this implies an average collapse stress of nearer 65% of yield. At the same time it will become increasingly difficult to achieve acceptably low fatigue stresses at details (see Chapter 13).

66 When an acceptable structure has been devised at all the critical sections (defined in Chapter 6) and the overall ship weight and bending moment distribution has been refined to the current stage of the design, there are still some important checks to be carried out, specifically on ultimate strength, torsional strength and vibration. Note that horizontal bending can usually be dismissed, as discussed in Chapter 3, because the bending moment is lower than the vertical figure while the strength in the horizontal plane is always greater than that in the vertical plane for conventional monohulls.

Ultimate Strength

67 So far only an estimate of the strength of individual panels of the structure has been made. It is necessary now to calculate with greater accuracy the panel strength and then to integrate these strengths to estimate the collapse load, known as the 'ultimate bending moment' (M_u), for the complete section.

68 The procedure for calculating the ultimate strength of a stiffened panel should be similar to that described by Smith (1977, 1981) and by Smith and Dow (1981), encapsulated into the computer program FABSTRAN (Dow and Smith - 1986). That is, for each combination of plate and stiffener a 'load shortening curve' such as those illustrated in *Figure 7.8* is to be produced, describing the elastic and plastic response of the panel up to the point where it is no longer capable of taking extra load. In preparing these curves the worst likely material properties (from Chapter 5 for example) should be used, but in association with moderate or typical initial deformations and

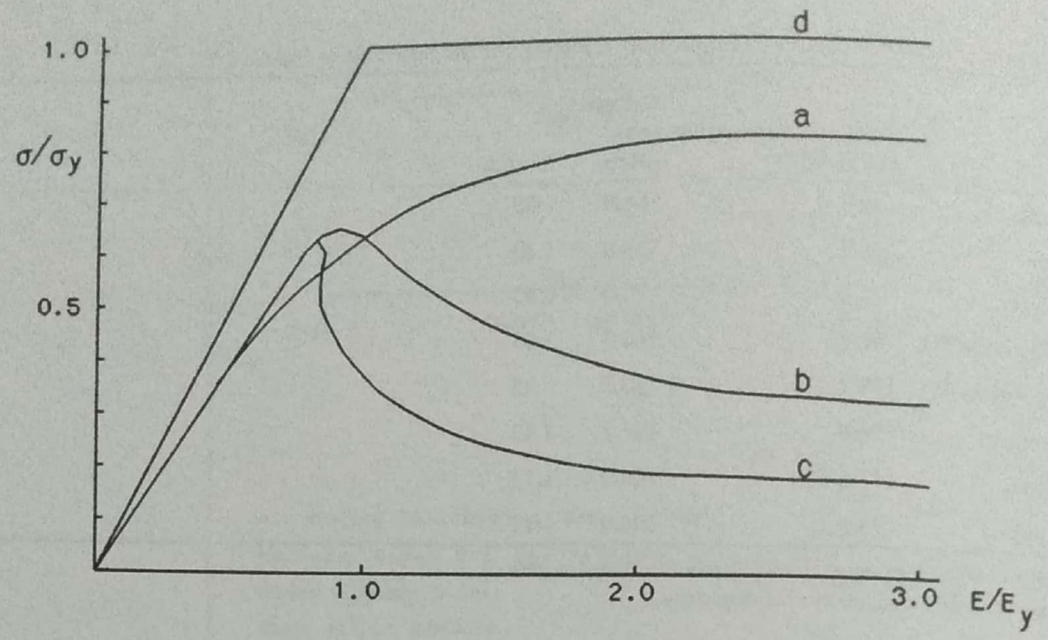

Figure 7.8 Typical Load Shortening Curves

residual stresses. The same boundary conditions as have been recommended previously during the synthesis process should also be used.

69 With reference to *Figure 7.8* the various direct types of curve should be noted. That marked 'a' can continue to absorb strain energy beyond its notional failure point (the point at which the maximum value of σ/σ_y is reached), although the stress cannot increase, and so such a structure will fail benignly as far as neighbouring panels are concerned. The curves marked 'b' and 'c' show a significant reduction in the ability of the structure to absorb strain energy after failure and such a panel will shed load to its neighbours, which in their turn may not be able to accept that load and the overall structure can collapse catastrophically. Clearly a response curve such as 'a' is the most desirable one, and it will usually be achieved by ensuring that the values of b/t and a/k are within the ranges quoted earlier (paragraph 5). Some typical load shortening curves for panels in *Figure 7.7* are illustrated in *Figure 7.9* and it can be seen that they are nearer to the shape of curve 'b' in *Figure 7.8* than to curve 'a', illustrating the effect of choosing b/t and a/k values higher than those recommended by Smith (1983) and discussed in paragraph 5. From these curves the values of the average failure stresses can be deduced and compared with those in *Table 7.4*. If the values are such as to invalidate the design, that is the margins against failure are no longer high enough, then the panels concerned will have to be redesigned.

70 Not all the structure will fail in the manner shown in *Figures 7.8 (a, b or c)* and *7.9*, some areas will have sufficient stiffness to resist premature buckling and will follow an elastic-perfectly plastic path as shown in curve 'd' in *Figure 7.8*. It is difficult to provide precise rules on which areas should fail in such a way, but they will usually be fairly obvious due to their stockiness, for example the keel structure in *Figure 7.7*. Another particular area which can be treated in this way is 'hard corners'; the near right angle corners between decks and the ship's side. If there is no clear extent for the corner structure apparent from the local scantlings, then a distance of 30t from the corner may be assumed.

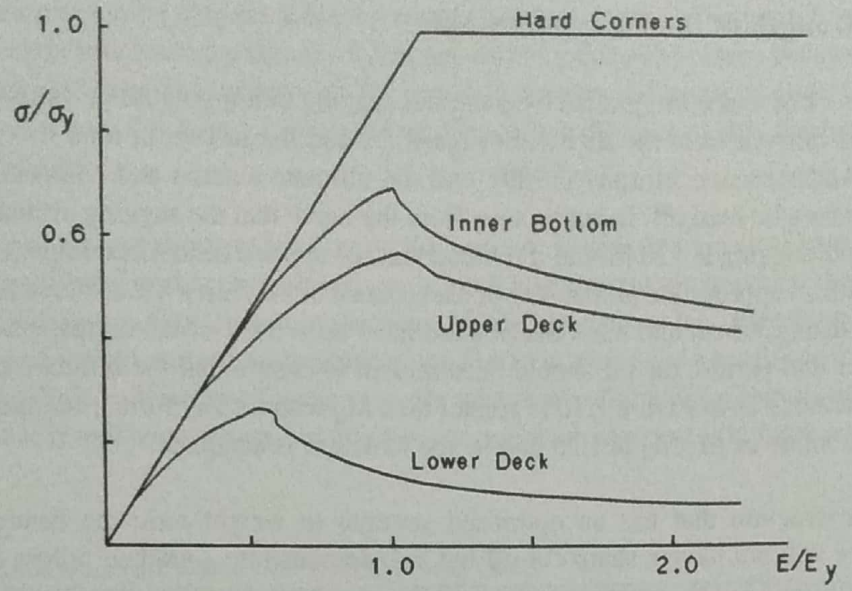

Figure 7.9 Examples of Load Shortening from Structure in Figure 7.7

71 When the longitudinal load shortening curves have been established for each element of the cross section of the hull, a complete bending moment/curvature relationship for the section can be calculated. This is best carried out using available computer programs such as NS94 from ARE Dunfermline described by Smith and Dow (1986), but may also be calculated directly using a simple procedure as follows and as given in more detail by Smith (1977):

a. The vertical curvature of the hull is assumed to occur incrementally with corresponding incremental element strains being calculated on the assumption that plane sections remain plane and that the structure bends about the instantaneous elastic neutral axis of the cross section.

b. Element incremental stresses are derived from incremental strains using the slopes of the load shortening curves (effective tangent moduli). In the case of unstable post buckling behaviour showing a re-entrant curve (that is, the strain *reduces* as stress increases for a short distance, as curve 'c' in *Figure 7.8*), the re-entrant portion of the curve has to be replaced with a short notional length with steep negative slope.

c. The position of the neutral axis is recalculated to allow for the change in the tangent modulus of each panel.

d. Element stress increments are integrated over the cross section to obtain bending moment increments. These, together with incremental curvatures which are derived from the original strain increments, are summed to provide cumulative values of bending moment (M) and curvature (φ).

It has to be said, however, that there is some conservatism built into the process because the dynamic whipping content of the loading means that the true ultimate failure will be higher due to inertial effects. An attempt to quantify the level of conservatism is presented by Dow et al (1981).

Ultimate Strength of the Design Example

72 Values of M and φ are plotted over the full sagging to hogging range as shown in *Figure 7.10*, which is derived from the section in *Figure 7.7* and the full set of load shortening curves taken from Anderson and Murphy (1989), and the ultimate tension and compression bending moments can then be read off. It can be seen from the curve that the sagging ultimate strength is 509 MN.m and hogging is 720 MN.m. Provided that the applied design bending moment (M_d) is calculated with an appropriate probability of exceedance in the ship's life as described in Chapter 3, and that a displacement and mass distribution have been used which correspond to the worst expected over that period, then it should be sufficient to ensure that the ultimate moment (M_u) from the curve such as in *Figure 7.10* is greater than M_d without any further margin. In this case the minimum value of M_u/M_d is 1.25 and so the structure is adequate.

73 For a structure that has an optimised strength to weight ratio the bending moment/ curvature plot will not have a sharp cut-off but will demonstrate a gradual failure analogous to curve 'a' in *Figure 7.8*. The curve in *Figure 7.10* shows a sharp cut-off because the structure so far designed has aimed at relatively high values of slenderness ratios so as to achieve widely spaced stiffeners and a *subjective* cost optimised arrangement. The weight per unit length of the section shown in *Figure 7.7* is 8.4t/m which compares with the weight of a typical modern frigate scaled to the same dimensions) of 6.5 t/m, in other words the penalty for the widely stiffened style of structure is a 29% increase in the weight of the *longitudinally effective* structure, or a little over 100 tonnes in a ship of this size. It is possible that the weight penalty would not be acceptable, and that a more weight efficient design would be pursued at the next iteration, but cost/weight trade-off studies would be needed to enable an objective conclusion to be reached.

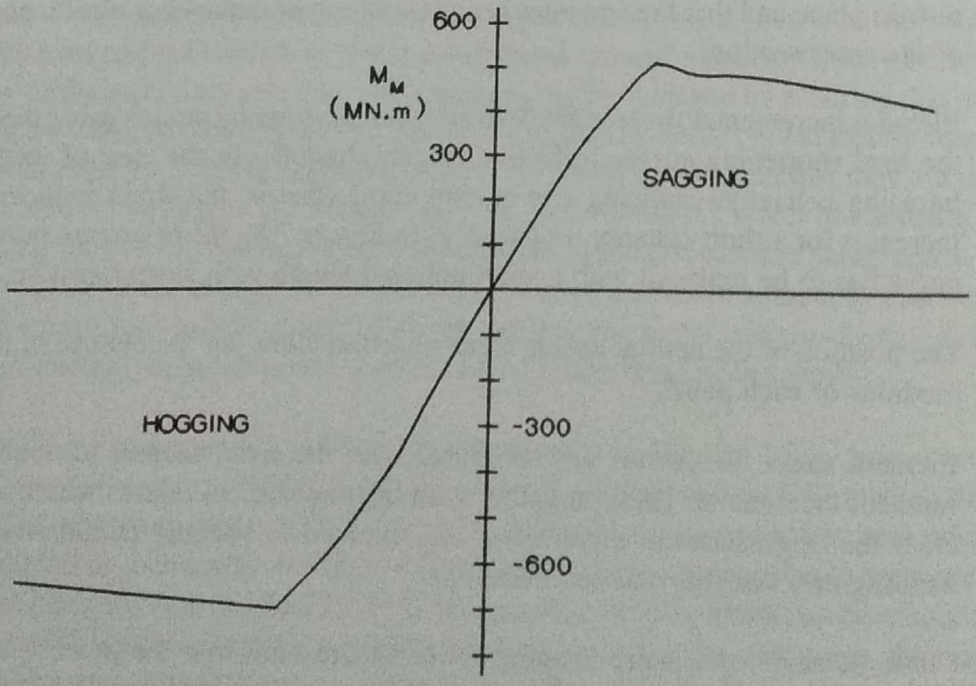

Figure 7.10 Bending Moment/Curvature Relationship for Example Hull Section

74 Another point to be noted is that for a shape of failure curve as shown in *Figure 7.10*, it is probably desirable to achieve a value of M_u/M_d somewhat greater than unity because of the rapid fall-off in the load carrying ability of the section, despite the conservatism mentioned in paragraph 71. A value of at least 1.1 is suggested, although in this case clearly the 1.25 achieved is more than adequate.

75 Satisfying the requirement for a minimum value of M_u/M_d at all critical sections is clearly a necessary condition for an acceptable structure, but it is important to remember that the ultimate strength calculations take no account of the effect of lateral pressure. It is, therefore, essential that for each panel of the structure the combined stress due to in-plane and lateral load is less than the collapse stress. This is illustrated for the example structure in *Table 7.5* where it can be seen that the requirement is met in each case but that there is not much margin for reduction of scantlings.

TABLE 7.5
Margins Against Compressive Collapse from Load Shortening Curves

	Applied Compressive Stress (MPa)	Collapse Stress by FABSTRAN (MPa)	Margin
Upper deck	149.1	165.6	1.11
Sheer strake	125.7	172.6	1.37
Internal deck	75.8	79.1	1.05
Ship's side above WL	52.2	159.4	3.05
Ship's side below WL	128.7 #2	135.7	1.05
Inner bottom #1	76.0	196.0	2.58
Outer bottom	147.5 #2	166.6	1.13
Vertical floors	93.5	127.7	1.37

#1 Margin will allow for tank working pressure (undefined as yet).
#2 Includes secondary compressive bending stress.

76 Comparison between the initial estimates of collapse stress given in *Table 7.4* and those derived from load shortening curves for the structure as designed, given in *Table 7.5*, shows that the initial estimates obtained from using the standardised load-shortening curves in the data sheets are mainly conservative. At only one area, the ship's side below the waterline, is the FABSTRAN figure more than 1% below the initial estimate. In the case of the ship's side, the effect of taking a true curved shape in *Figure 7.7* has been to increase the longitudinal stiffener spacing locally, and a recalculation using the initial method but with the correct spacing suggests a collapse stress of 128.3 MPa which is then less than the FABSTRAN result. There is no clear correlation in the differences between the two methods, but there is a trend towards the initial method being more conservative at higher slenderness ratios, illustrated particularly by the internal deck which was not acceptable in *Table 7.4* but is just adequate in *Table 7.5*. The very low collapse strength of this relatively unimportant part of the structure demonstrates the inefficiency resulting from the high slenderness ratios for that particular arrangement, that is b/t = 149 and a/k = 134, but the widely spaced flat bar stiffening should be very cheap to build.

77 The design method illustrated shows the procedure to be adequately conservative in the limited range of scantlings used. However, while it is recommended for use it should be noted that there are a number of unexplored areas which merit further work. In paragraph 5 it is suggested that that the limitations on the values of slenderness ratios recommended by Smith (1983) were too restrictive, and the example uses a wider range. This aspect clearly needs deeper study leading to a discussion on the sensitivity of ultimate strength to variations in ß and λ in different parts of the structure. Currently only the subjective view can be presented, that it will be more important to keep within the suggested ranges in structure furthest from the neutral axis. It is also worth noting that structure designed to most weight efficient is not necessarily the cheapest to build, as discussed by Chalmers (1986). The example chosen was aimed intuitively to be relatively cheap, with thick plate and widely spaced stiffeners, but there is an indication from the shape of the load-shortening curves that the strength could be improved by a rearrangement of the scantlings. However, if the cross section area of the steel is reduced to save weight, the average stresses will increase and so safety margins will be squeezed.

78 A second point to be noted is that average imperfections have been assumed throughout. If lighter plate, or indeed heavier plate, is used then more or less severe imperfections may result. This opens the question of sensitivity to imperfections and this aspect would be accounted for in further iteration of the design.

79 The question of lateral pressure has been discussed briefly (paragraph 47) and a simplifying assumption made that the stresses due to bending may be added to the direct primary compressive stresses. Smith, Anderson et al. (1991) have presented curves illustrating the effect of lateral pressure on collapse, and although a superficial inspection suggests that the difference between the simplified procedure presented here and the procedure of the reference is small, it is recommended that a more rigorous check is undertaken for structure with significant lateral pressure.

80 Another variable is the ratio of stiffener area (A_s) to total area of plate plus stiffener (A). The column collapse curves (*Figure 7.5*) are drawn for $A_s/A=0.2$ which is a conventional typical value. Smith, Anderson et al. (1991) present curves for the effect on strength of changes in A_s/A from 0.1 to 0.4, and while the effect does not appear to be very significant, there is some sensitivity especially for low λ and high ß.

Torsion

81 Traditionally torsion has not been seen as a problem except in ships with very large deck cut-outs such as deep hold container vessels, or for ships of shallow section such as 'Great Lakers'. However, in recent years two problems have become apparent which can be ascribed to lack of torsional strength or stiffness. The first of these concerns ships which are required to tow sonar arrays and which during towing operations have to proceed at both low and high speeds into quartering seas. Such operations can result in comparatively large torsional loads on the hull which have not previously been allowed for in design and for which there is no precedent. Even so, it would probably not be important if it was not for the fact that modern gas turbine propelled warships have much larger holes in their main longitudinal structure than their steam driven forebears for intake and exhaust, and are therefore in some respects verging on the structural arrangements of a container ship. Furthermore, even in steam ships the torsional stresses generated can lead to unexpectedly high stresses at details, and fatigue cracking in areas where the designer would not have predicted.

82 The second problem concerns torsional vibration, again in ships with large cut-outs, where coupled torsional and horizontal bending resonances can be excited by the propulsion train as discussed by Bishop et al (1986). If operational or living spaces are positioned in way of anti-nodes of the mode shape, and especially if they are distant from the notional centre of rotation of the torsional vibration, the large linear displacements encountered can be both irritating and disruptive.

83 In consequence, it may be desirable to estimate the torsional strength and stiffness of the hull both to predict frequencies and mode shapes and to avoid areas of high stress due to torsional effects. There are, unfortunately, no simple rules, although some guidance on torsional vibration aspects is given in Chapter 14. A method for calculating shear stiffness of simple sections is presented by Bhat and de Oliviera (1985). The following paragraphs are a summary of the full calculations necessary which may be complex and require computer analysis; references are supplied for further detail and it will be seen that the majority of these are derived from aerospace developments where torsion is an overriding problem.

84 The governing equation relating applied torque (τ) to twist (ϑ) is

$$\mathrm{T} = GJ\partial\vartheta/\partial x - E\Gamma\partial^3\vartheta/\partial x^3 \tag{7.15}$$

where G, E are shear and Young's modulus
J is torsional constant (polar second moment for axisymmetric sections)
and Γ is warping constant.

The first term in equation 7.15 represents the rate of twist due to pure torsion, sometimes known as St Venant torsion, and leads to shear stresses in the cross section. The second term represents the rate of twist due to warping when the cross section does not, in general, remain plane and points on the section are displaced axially as discussed for simple sections by Murray and Attard (1987). If the cross section is constrained, either by a local discontinuity or by an external support, then axial direct stresses as well as shear stresses are set up.

85 To estimate the stresses due to a given torque, two activities are clearly necessary. First the two geometric constants, J and Γ, need to be calculated and, secondly, the relationship between shear stress and rate of twist needs to be established. However, for the majority of ship structural sections, unless there are very rapid changes in torsional stiffness, the value of $\partial^3\vartheta/\partial x^3$ is very much less than $\partial\vartheta/\partial x$ and so the warping effect may be neglected. Chalmers (1980) has shown that omitting the warping effect underestimates the torsional stiffness by about 10% in a closed or restrained section, but that this is at least in part compensated for by the imperfections in a real structure. For further details of the theory for thin walled structures, and especially for warping effects, see Argyris and Dunne (1947), Cheung and Koo (1988) and Koo and Cheung (1988), and textbooks such as Megson (1974).

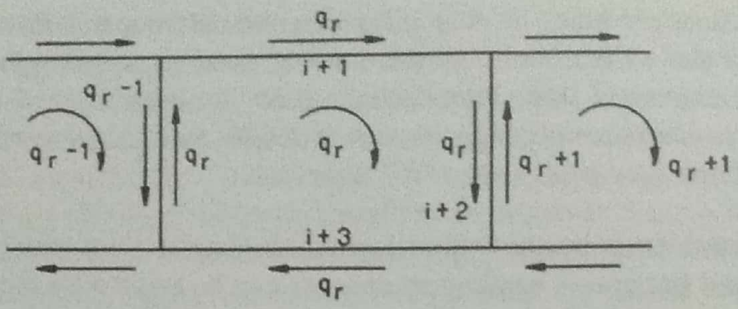

Figure 7.11 Shear Flow Loops

86 Returning to the problem of pure torque, consider a structure consisting of a general set of loops around which continuous shear flow (defined as $q = \tau t$, where τ is shear stress and t is thickness) can exist, as shown in *Figure 7.11*. Neglecting warping, each loop must have the same rate of twist Θ ($= \partial\vartheta/\partial x$) which is defined as

(7.16)

$$\Theta = \frac{1}{2A_r G}\oint_r (q/t)\, ds$$

where A_r is the area enclosed by the r^th loop
q is the shear flow constant around each loop
t is the wall thickness
and $\oint_r ds$ indicates integration around the walls of the r^th loop.

87 If, for the r^th loop shown, the walls are identified i to i+3 and if, along each wall i, δ_i is defined as $\int(1/t)\, ds$, then from equation 7.16

(7.17)

$$\Theta = \{\delta_i (q_r - q_{r-1}) + (\delta_{i+1} + \delta_{i+3})\, q_r + \delta_{i+2}(q_r - q_{r+1})\}/2A_r G$$

This equation can be rearranged for the whole structure in matrix form as

(7.18)

$$2\Theta G\underset{\sim}{A} = \underset{\sim}{\delta}\underset{\sim}{q}$$

where $\underset{\sim}{A}$ is the $1 \times N$ matrix of loop areas (for N loops)

$\underset{\sim}{q}$ is the $1 \times N$ matrix of shear flows around the loops

and $\underset{\sim}{\delta}$ is the $N \times N$ matrix of the geometric integrals around the loops.

It will be noted that the diagonal terms in $\underset{\sim}{\delta}$ are equal to $\Sigma(s/t)$ for all the walls of length s in the loop, and the off diagonal terms are equal to $-\Sigma(s/t)$ for the walls separating adjoining loops; the off diagonal terms are zero where loops are not contiguous.

88 By inverting $\underset{\sim}{\delta}$, $\underset{\sim}{q}$ may then be found from

(7.19)

$$\underset{\sim}{q} = 2\Theta G\underset{\sim}{\delta}^{-1}\underset{\sim}{A}$$

The torque carried by each hoop is $2A_rq_r$ and so the total torque carried by the whole structure is given for N loops by

(7.20)
$$T = \sum_{r=1}^{N} 2A_rq_r$$

If unit torque is applied with a consequent rate of twist Θ_1 and a 'normalised' shear flow q_r^* in each loop is defined such that $q_r = 2\Theta_1 G q_r^*$, then q_r^* is a geometric parameter (with dimensions of area) equal to the elements of $\underset{\sim}{q}^*$ from

(7.21)
$$\underset{\sim}{q}^* = \underset{\sim}{\delta}^{-1}\underset{\sim}{A}$$

whence

(7.22)
$$2\Theta_1 G = 1/(2\sum_{r=1}^{N} A_rq_r^*)$$

and the shear flow in each loop due to unit torque can be derived directly from the definition of q_r^*. The shear flow in each *wall* can then be found by linear superposition as indicated in *Figure 7.11* and by multiplying the result by the actual torque T.

89 The torsion constant is defined from equation 7.15 such that $J = T/\Theta G$. Again taking unit torque it will be seen that $J = 1/\Theta_1 G$ or

(7.23)
$$J = 4\sum_{r=1}^{N} A_rq_r^*$$

90 There is now sufficient information to calculate, somewhat tediously, the shear stress around any closed section, consisting of multiple loops, due to an applied torque. In fact, the structure illustrated in *Figure 7.7* is not well able to resist torsion because the shear flow can only exist around the double bottom. For adequate torsional stiffness a longitudinal bulkhead should be introduced between 1 and 2 decks alongside the centreline hole to provide another shear flow loop at the top of the structure.

91 Suitable margins must be in doubt for ship type structures as the calculation has only rarely been carried out and is not well documented, but as an unsubstantiated recommendation it is suggested that, using the torque recommended in paragraph 64 of Chapter 3 (which is not an extreme load) then the torsional stresses derived above should not exceed 30% of the shear yield stress. Should this criterion not be met then the structure is not necessarily unacceptable but further consideration should be given to the distribution of material around the section to resist torsion.

92 If vibration calculations are to be carried out then it will also be necessary to find the shear centre, which itself involves calculating the effective shear areas for shear force in the y and z directions. These calculations are even more complex and are discussed in Chapter 14 with the mathematical details to be found in Bishop et al (1986).

Explosive Loading

93 All of the foregoing discussion has concentrated on wave loading. Warships also have to withstand explosive loading and, while this requirement does not normally dominate the overall structural design, it can affect the arrangement of details. In the hull this involves particularly the support to main transverse bulkheads, which is discussed in Chapter 9, and resistance to whipping which is covered in Chapter 15. In the case of mine countermeasures vessels, which are designed to withstand underwater explosions during normal operations, and which may not be built of ductile materials, consideration of the whipping response is an essential part of the analysis of longitudinal strength.

Summary of Procedures and Recommended Safety Margins

94 It will have been noted that the procedures discussed above are distinctly interactive and iterative, but the following paragraphs are an attempt to summarise the process as a form of aide memoire for the designer. While the methods are divided into categories, for example design against in-plane load or lateral pressure, checks against *all* loads on *all* elements of the structure must be carried out. (Note that in the example presented above these have not been rigorously completed both in the interest of brevity and to reduce the extent of repetition.) Some checks are also required on combinations of loads (paragraphs 96-97).

Initial Synthesis

95 From the geometry of the section, and the allowable stresses, estimate the mean thickness of material required in each structural 'panel' to resist the design bending moment (paragraphs 5-7). Calculate the direct bending stress at the worst point on each panel.

96 For each panel decide what will be the critical loading (in-plane, lateral, shear etc) and commence detailed synthesis using that load. It will usually be best to start the design with the upper strength deck which is in the most critical area and then to use the stiffener spacings derived there as a guide to spacings elsewhere in the structure. All spacings must be consistent across boundaries between panels.

In-Plane Load

97 a. Match stiffener size and actual plate thickness (allowing for rolling and corrosion reductions) to the mean thickness required (paragraph 23).

b. Estimate average in-plane compressive strength (σ_a) for the stiffener furthest from the neutral axis (paragraphs 16, 38-39). Ensure that σ_a is greater than the applied bending stress.

c. Estimate transverse stiffener spacing (and size) using lateral load as a criterion if appropriate, or a suitable balance between longitudinal and transverse stiffener sizes otherwise (paragraph 41).

d. Check overall buckling collapse (paragraphs 17, 42).

e. Check longitudinal stiffeners against tripping (paragraph 43); critical in-plane tripping stress should be at least 30% higher than the critical in-plane buckling stress for the stiffener and attached plating (paragraph 18).

Lateral Pressure

98 a. For appropriate boundary conditions estimate the minimum section modulus for combinations of stiffener and effective breadth of plate (paragraph 48); for pressure alone the highest applied bending stress should be 20% less than the yield strength for frequent loads, and 10% less for rare (test) loads (paragraphs 23, 48).

b. Match an appropriate stiffener and plate to the required modulus (paragraph 48).

c. Check that there is a margin of at least 2.0 between shear stress in stiffener webs and the shear yield stress (paragraph 24).

d. Check against tripping under lateral load (paragraph 49). Tripping critical stress should be at least 4 times the applied bending stress for primary structure and 3 times for other structure (paragraph 25).

Shear Loads

99 a. Calculate the applied shear stress (τ_d) on the worst plate panel and the worst stiffener (paragraph 29).

b. Check plate shear strength (τ_c) and ensure $\tau_c \geqslant \tau_d$ (paragraphs 28, 58).

c. Check panel shear strength (τ_{sc}) and ensure $\tau_{sc} \geqslant \tau_d$ (paragraphs 30, 31, 58).

Combined Loads

100 For a combination of in-plane and *continuously* applied lateral load, for example on the outer bottom, the *sum* of the local *compressive* bending stress and the applied in-plane stress must be less than the *average* in-plane collapse stress of the panel (paragraph 47).

101 For a combination of in-plane and shear loads, for example in the shear strake, the safety factor between the total (in-plane plus local bending) compressive stress and the average in-plane collapse stress should be increased to be not less than $\tau_{sc}^2/(\tau_{sc}^2 - \tau_d^2)$ (paragraph 61).

Ultimate Strength

102 a. Calculate load shortening curves for all stiffener-plate combinations including 'hard corners' (paragraph 68).

b. Compare panel failure stresses with earlier estimates (paragraph 89b) and redesign if necessary (paragraphs 69, 75).

c. Calculate the ultimate strength of the section and ensure that it is greater than the design bending moment (paragraphs 71-72).

Torsion

103 If the section has significant cut-outs extending over some length of the hull, or if there are few shear flow loops, carry out a torsional strength calculation (paragraphs 82-85) and ensure that shear stresses are not more than 30% of the shear yield stress (paragraph 87).

Whipping

104 Carry out a whipping analysis if necessitated by the operational scenario of the ship, as described in Chapter 15 (paragraph 32 et seq.).

paragraphs

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